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Special Features of the Behavior of the Liquid Phase in High-Speed Compressors of Aircraft Engine-Derivative Gas-Turbine Units and Their Impact on Characteristics and Effectiveness of "Wet" Compression

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Abstract—The paper deals with the processes of "wet" compression with injection of water into axial-flow compressors of aircraft engine-derivative gas-turbine units, which are characterized by a relatively high speed of rotation and small standard dimensions. Given for two typical machines are the results of analysis of the behavior of water films on the surfaces of stator and rotor blades at different compressor stages (including the separation of moisture to the casing) and their impact on the characteristics and effectiveness of the process of "wet" compression.

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High interest shown today in "small-scale" and autonomous power generation defines the promise held by the use of low-power gas-turbine units (GTUs) based on aircraft engine-derivative gas-turbine engines. In so doing, much urgency is given to the problems of raising efficiency η_{GTU} and specific power N_{sp}, which define the economic feasibility and the payback periods of such units.

One of the efficient and low-cost methods of raising η_{GTU} and N_{sp} of aircraft engine-derivative GTUs is by injecting small amounts of water (1–3% of the airflow rate) into the compressor for reducing the temperature of air being compressed during the evaporation of water. The theoretical (of which [1, 2] must be specially noted) and experimental [3, 4] papers dealing with the investigation of "wet" compression in GTU compressors indicate the possibility of an appreciable increase in η_{GTU} of compressors and of a certain increase in the compression ratio, in the flowrate of the working medium, and in the stability of operation of the machine. At the same time, the efficiency and specific power of the GTU as a whole increase significantly (especially, in the case of combined cycles, including operation with "wet" compression and heat regeneration [6]) and nitrogen oxide emissions with stack gases decrease significantly [3].

Analysis of the studies cited above leads one to conclude that the process under consideration is calculated well for machines of relatively high power and not too high a speed of rotation (n = 3000–5000 rpm). At the same time, as regards real aircraft engine-derivative machines, some problems associated with the injection of water call for further detailed study.

The main features of axial-flow high-speed compressors of aircraft engine-derivative GTUs of low and moderate electrical power (from several hundred kilowatts to 20 MW) and the peculiar behavior and evaporation of water being injected, which is defined by these features, are as follows:

(i) the relatively small distances between neighboring blades and the large relative surface of these blades define the intensive separation (capture) of the bulk of the droplets to the rotor blades of each stage and the prevalence of evaporation of moisture from the surface of films forming on the blades (as a rule, the fraction of evaporation from the surface of droplets does not exceed 20%);

(ii) the high speed of rotation of the compressors of small-size aircraft engine-derivative machines produces relatively high centrifugal and Coriolis forces that affect strongly the behavior of films of water on the rotor blade surfaces; and

(iii) the small width of the blades defines the relatively high friction forces of airflow on the surface of water films.

Based on the results of analysis and on the results of foregoing experimental and theoretical investigations of "wet" compression in GTU compressors (first of all, of [1]), the calculations were performed using a physical model of the behavior of the liquid phase in the compressor flow path that takes into account the special features of small-size aircraft engine-derivative GTUs. The characteristic features of this model are as follows:

(i) almost all of the water droplets dispersed by atomizers and delivered to the compressor stage are

separated on the blade surface within the stage and form films that move under the effect of centrifugal forces in the radial direction, and under the effect of friction forces of air flow in the longitudinal direction;

(ii) because of the relatively high values of the forces identified above, the film thickness on the blade surface turns out to be very small (estimated at 10–12 μm) and, accordingly, secondary droplets sized 8–10 μm are formed during the detachment of droplets from the blade edges;

(iii) because of their small size, the droplets obtained as a result of such fragmentation are entrained by the flow in the longitudinal direction to the guide-vanes row; however, on the next rotor wheel, they are separated again on the blade surface; and

(iv) the liquid phase evaporates from the surface of both droplets and films on the stator and rotor blades (in the latter case, only on the concave side), and the heat transfer to gas flow is taken into account on both sides.

By and large, the calculation algorithm is close to those developed in [1, 2, 7–16] and includes:

(i) one-dimensional gasdynamic calculation of flow of a steam–air mixture on the mean over the cross-sectional diameter of the compressor flow path with an account of the variable flow rate of the working medium in each stage similarly to [8];

(ii) calculation of the evaporation of droplets and water films on the stator and rotor blades in a moving steam–air mixture;

(iii) two-dimensional calculation of the motion of droplets in a steam–air mixture in the plane coinciding with the plane of mean cross section of the blade channel according to [9];

(iv) the calculation of the thermophysical properties of the steam–air medium as a mixture of perfect gases;

(v) analysis of the motion of the water film on the rotor blades under the effect of centrifugal force and friction forces and of the motion of the film on the guide vanes and the casing under the effect of friction forces of gas flow with the positive pressure gradient along the flow path, which is characteristic of the compressor, according to [10]; and

(vi) analysis of the size reduction by the gas flow of the water film separated from the trailing edges of the blades, similarly to as in [11].

The calculation of an axial-flow multistage compressor is performed by stages using the known geometric dimensions of the channel, the known compressor parameters (airflow rate, compression ratio, speed of rotation), and the known (for the preassigned operating mode) stage distribution of the total specific energy (head) and isentropic efficiency under conditions of operation without delivery of water.

The basic equations for the calculation of the thermodynamic parameters of gas in the stage are provided by the equations of energy and work under nonisentropic adiabatic conditions, the detailed notation of which is given in [12].

In view of the weak intensity of the runoff of steam from the interface (steam "injection"), the heat transfer between a droplet and gas flow is described by the dependence of the Nusselt number of the droplet on the Reynolds number under conditions of its relative motion in an air–steam flow [13] (Marschall–Ranz dependence),

$$\mathrm{Nu}_d = 2 + 0.6\sqrt[3]{\mathrm{Pr}}\sqrt{\mathrm{Re}_d}K_{t1}.$$

The heat transfer between a thin water film on the blade and the gas flow past the blade is expressed by the formula similar to that for the Nusselt number on a flat plate,

$$\mathrm{Nu}_f = 0.032\mathrm{Re}_f^{0.8}\mathrm{Pr}^{0.43}K_{t2}.$$

Here, K_{t1} and K_{t2} are the coefficients that allow for the flow turbulence level and, in the case of heat transfer to the film, for the presence of a positive pressure gradient. The estimation of K_{t2} for the characteristic conditions of flow in a TV-3-117 compressor (typical of low-power aircraft engine-derivative machines) was carried out by Reviznikov using a procedure similar to that given in [14]. These calculations revealed that, because of the effect of two oppositely directed factors, the value of K_{t2} under conditions being considered is close to unity.

The surface temperature of evaporating droplets and films was determined in accordance with [15] by the relation derived from the conditions of similarity of the processes of heat and mass transfer and confirmed experimentally,

$$T_f = T_0 - \frac{r\mathrm{Le}_b}{\bar{c}_p}\frac{x_b - x_0}{1 - x_b},$$

where T_0 is the air temperature, r is the heat of evaporation, $\bar{c}_p$ is the specific heat capacity of the steam–gas mixture, Le_b is the Lewis number with the parameters of the boundary of phase transition, and x_0 and x_b denote the mass fraction of steam in the air in the main flow and on the interface at temperature T_f.

The viscous forces in the film, as well as the forces of friction of air flow on the film surface and the centrifugal mass forces acting on the films of the rotor blades, were taken into account in considering the motion of films on the surface of the stator and rotor blades. The balance of water in the films included the delivery of moisture with droplets and its loss due to evaporation. Entrainment of droplets from the surface of films of the rotor blades was ignored (as distinct from in [7, 10]), because of the presence of rather high Coriolis forces pressing the films against the blade surface. Indeed, the Coriolis acceleration proportional to ωv_y (ω is the angular speed of rotation, and v_y is the radial velocity of film flow) may be of the order of 10^3g for the condi-

Table

Characteristic	TV-3-117	AL-21F3
Reduced flow rate of air under rated conditions, kg/s	7.78	91 (86.1*)
Number of compression stages	12	14
Rated reduced speed of rotation (at point η = max), rpm/min ($\bar{n}$ = 0.89)	17950	7500 (7320* $\bar{n}$ = 0.870)
Compression ratio π_{comp}*	7.65	13.8 (11.0–11.5*)
Adiabatic η_{comp}	0.820	0.856 (0.835*)
Mean diameter D_m of first-stage rotor blade, mm	328	870

* Parameters of the modes considered in this paper.

tions of TV-3-117 compressor. This "pressing" of films to the blade surface causes a significant increase in their stability to wave generation and entrainment of droplets. In the absence of entrainment of droplets, the thickness of films and the velocity of their motion increase appreciably (severalfold). As a result, the radial drift of moisture to the compressor casing increases significantly. This is confirmed to some extent by the data given in [7]. The absence of entrainment of films and reflection of droplets from the blade surface results in a shift of moisture toward the casing increasing abruptly from 2–4 to 40% even at the first stages of compression.

The motion of moisture being separated along the surface of the compressor casing was treated as the motion of films acted upon by the forces of friction of airflow in addition to the forces of friction against the casing surface. Also taken into account was the tangential momentum due to the delivery of droplets separated from the end edges of rotor blades.

The results of calculations of the behavior of moisture in TV-3-117 and AL-21 compressors (which we performed) and in a GT-009 compressor [16] lead one to assume that, because of the effect of centrifugal forces, the largest (over 60–80%) fraction of moisture must be separated on the casing even in the first several stages of compression. However, this statement fails to fully agree with the experimental data. At any rate, no information is contained in [3, 4] about such intense separation of moisture on the casing. The pattern of moisture behavior is apparently more complex, and the effect of certain mechanisms causes a significant part of water delivered to the compressor casing to return to the vanes. The following such mechanisms may be identified: first of all, part of the films separating from the ends of rotor blades disperses to fine droplets, fails to reach the casing surface, and returns to the compressor flow path under the effect of turbulent pulsations; second, as distinct from the films on the rotor blades, the films on the casing surface are subjected to more intense wave generation and the entrainment of droplets appears to be very real; and, finally, during their motion on the casing surface, the films run against the roots of the guide vanes and either separate or move on the guide vanes to the compressor axis and are separated by the flow.

The "wet" compression and the behavior of the liquid phase in the flow path of high-speed axial-flow compressors of aircraft engine-derivative gas-turbine units were calculated for TV-3-117 (GTE-1500) [17] and AL-21F3 [4, 18] compressors, the basic characteristics under rated conditions of which are given in the table.

Check calculations of operation modes without delivery of water were performed for both compressors, as well as calculations of modes of "wet" compression and of the characteristics of water films on the surfaces of the rotor and stator blades in stages of compression and of films on the internal surface of the casing. For estimating the effect made by individual factors, respective calculations were performed both disregarding the centrifugal forces and in view of them. In the latter case, options were considered without the moisture returning from the casing surface to the guide-vane surface, as well as with a part of the moisture returning, in accordance with the last of the mechanisms mentioned above (partial flowing down of films from the casing to the guide vanes). The impact made by these factors on the liquid phase balance, first of all, on the degree of evaporation (because of the variation of wetted blade surface), as well as on the moisture balance at the compressor outlet for different relative values of water flow rate for injection, is given in Figs. 1 and 2.

The centrifugal forces lead to a significant variation of the distribution of moisture (see Fig. 1). In particular, the fraction of moisture on the casing reaches values of 0.45 for AL-21F3 and 0.48 for TV-3-117, and the fraction of evaporated moisture decreases from 1.0 to 0.52 for AL-21F3 and from 1.0 to 0.45 for TV-3-117.

The return of water from the casing surface causes a significant variation of the distribution of moisture. For example, the integral flow of water to the casing with 50% return at each stage decreases in AL-21F3 from 14 to 8% in the first two stages of compression and from 48 to 18% at the compressor outlet. In spite of this, the integral fraction of evaporated water varies insignificantly (see Fig. 1a). Therefore, the inclusion of return of moisture from the compressor casing to the blades, which is made with a view to obtaining a more complete description of the process being analyzed, introduces no fundamental changes into the final results of calculations (at least, for AL-21F3 and TV-3-117 compressors).

As the flow rate of water for injection x_{water} is increased from 0.5 to 2.0–2.5% of the airflow rate, the amount of evaporated water increases from 0.06 (with respect to the air flow rate) to 0.44% for TV-3-117 (at x_{water} = 2.0%) and from 0.34 to 1.1% for AL-21F3 (at

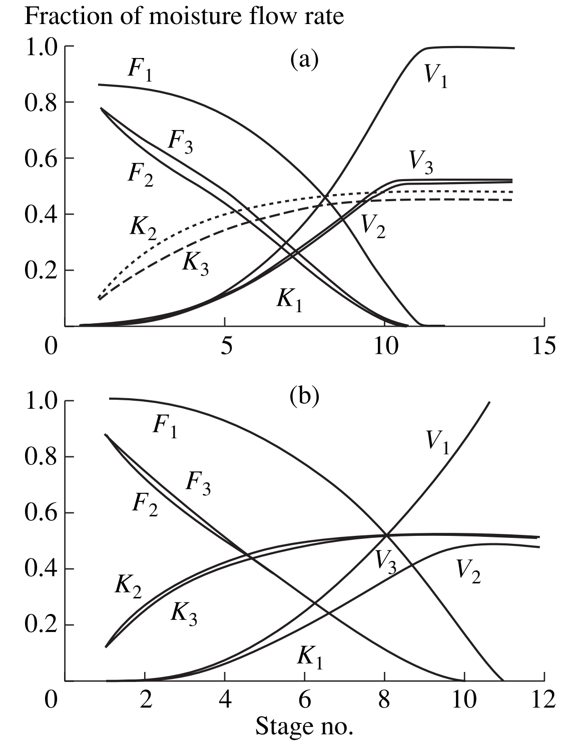

Fig. 1. The distribution of relative fractions of injected water in films on the blades (F_1–F_3), in steam (V_1–V_3), and on the casing (K_1–K_3) in the compression stages of compressors (a) AL-21F3 and (b) TV-3-117, x_{water} = 2%. The subscripts indicate: 1—ignoring the centrifugal force, 2—in view of the centrifugal force without return of the film to the guide-vane surface, and 3—in view of the centrifugal force with partial return of the film to the vanes.

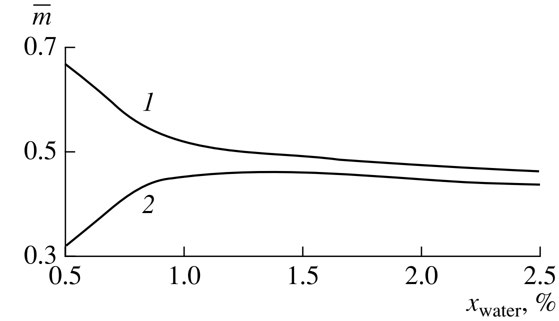

Fig. 2. The moisture balance at the outlet from AL-21F3 compressor as a function of the amount of injected water related to the air flow rate: (*1*) fraction $\overline{m}_{evap}$ of evaporated water; (*2*) fraction $\overline{m}_{cs}$ of water on the casing.

x_{water} = 2.5%). In so doing, the fraction of evaporated water varies nonlinearly with increasing delivery of water to the inlets of compressors (see Fig. 2). This is apparently associated with the special features of formation and distribution of water films on the compressor surfaces, including the emergence and growth of "dry" surface regions on the blades of the last stages (Fig. 3).

Analysis of the calculated characteristics of film flows reveals that the uniform radial distribution of the longitudinal and radial components of velocities of flow of films and their thickness is observed only at the first (after the point of injection) stage of compressor. In the subsequent stages, the fast-growing maxima of film flow velocities are located in the vicinity of the end face edges of the rotor blades. The longitudinal velocity of film flow on the surface reaches at its maximum a value of 20 m/s. "Dry" regions emerge at the roots of the rotor blades, with their sizes increasing from stage to stage to cover more than half the entire surface at the last stages.

A similar pattern of distribution on the blade surface is observed for radial velocity v_y as well; in so doing, its values of 15–20 m/s at the first stages of compression are maximal and decrease to 0.8–1.0 m/s at the last stages, apparently because of the appreciable decrease in the thickness of film on the blades of these stages (Fig. 4).

The film thickness in the vicinity of the end faces of the rotor blades increases along the gas flow; at the trailing edge, this thickness increases uniformly throughout the height of the blade (up to its end face) to 10–25 μm.

For the given values of film thickness at the trailing edges of the rotor blades (approximately the same values of film thickness are observed at the edges of the guide vanes), the secondary droplets formed during the atomization of the films separating from the edges are 8–16 μm in size. Under conditions typical of small-size compressors with a high speed of rotation, it is the entrainment of films from the trailing edges of blades (both rotor and stator blades) that forms secondary droplets by the known mechanism of collapse of a thin sheet of liquid in gas flows [9].

The distributions of the longitudinal velocity v_x of the flow of films on the guide vanes are similar to the v_x profiles on the rotor blades located upstream of the

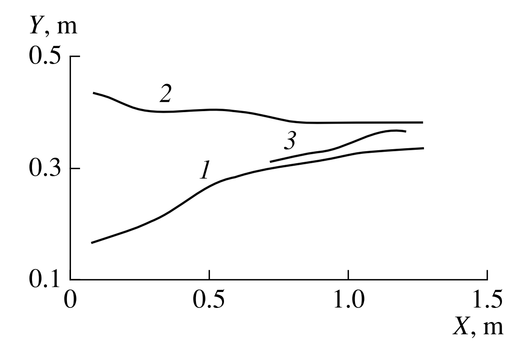

Fig. 3. The fraction of "dry" surface on the rotor and stator blades of AL-21F3 compressor, x_{water} = 2.5%: (*1*) coordinate of the sleeve surface, (*2*) coordinates of the casing generatrix, (*3*) upper boundary of dry region; *Y*—coordinate counted from the compressor axis, *X*—longitudinal axis of the compressor.

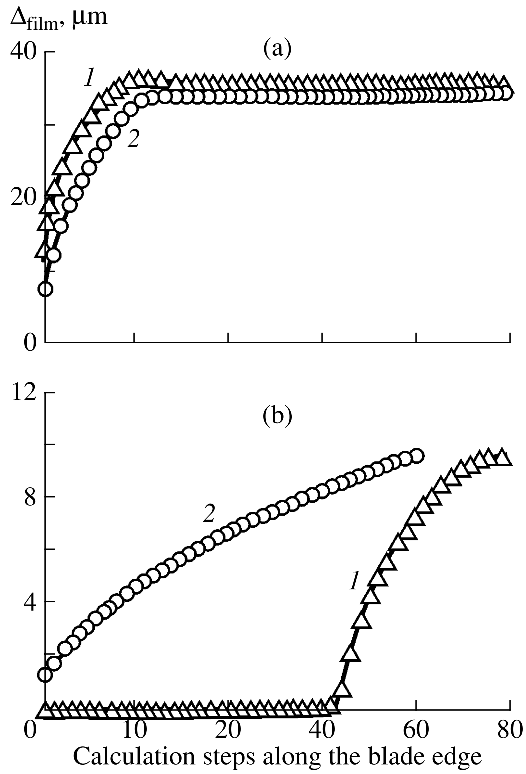

Fig. 4. The thickness of films on the rotor blades of the (a) 1st and (b) 12th stages of AL-21F3 compressor, x_{water} = 2.5%: (*1*) film thickness at the trailing edge (50 steps over the blade height), (*2*) film thickness at the upper end edge (40 steps).

flow. However, as distinct from the films on the rotor blades, the variation of v_x in the direction of flow on the guide vanes is relatively small.

The radial increase in film thickness and the presence of maxima of velocity and film thickness in the vicinity of the blade end face is caused by the combined effect of the radially increasing centrifugal forces and concentration of droplets in the gas flow that is incident on the blade.

The results of calculations of "wet" compression performed for the AL-21F3 compressor (π_{comp}, η_{ad}, and the temperature of the medium) were compared with the data of experiments performed at the enterprise MMPP Salyut and the Central Institute of Aviation Motors (TsIAM) [4, 18]. The reduction of temperature of air being compressed ΔT_{out} at the compressor outlet in experiments with x_{water} = 2.5% and reduced speed of rotation $\bar{n}$ = 0.87 was approximately 68 K [4]. The calculated reduction of average air temperature at the compressor outlet was 33 K. The difference between the values of ΔT_{out} may be due to the presence in the compressed airflow of an appreciable amount of unevaporated droplets of water that reached the thermocouple junction and caused a significant decrease in its temperature; this possibility must be taken into account in performing experiments and assessing measurement results. An increase in the overall compression ratio in a compressor as a result of injection of water at its inlet is observed both in experiments and in calculations; however, concrete comparison is difficult because of incomplete agreement between similar modes of compression and a very high "steepness" of the characteristics of compression π_{comp} (G_{air}) for a fixed value of $\bar{n}$.

We should note that the results of calculations of the behavior of the liquid phase, as well as of the integral characteristics of a compressor, depend rather strongly on the assumptions made in the calculation model. Therefore, in view of the limited availability of experimental data (which serve as the test base for any calculation procedure), the calculation results must be treated first of all as the reflection of certain tendencies to be checked experimentally. In particular, this is true of the inference of the weak effect of the initial air temperature (up to 50°C) on the characteristics of "wet" compression of high-speed machines, of the expediency of injection of water into the zone of π_{comp} = 1.6–2.2, and of certain independence of the final results on the degree of dispersion and distribution of the injected moisture over the channel height (naturally, up to a certain limit).

The process of compression of the working medium in a compressor is characterized by an ideal work of compression L_0 and internal relative efficiency. The latter parameter indicates the degree of perfection of the machine and is determined by the amount of energy loss in the machine. In the absence of heat transfer to the environment, the internal relative efficiency is usually provided by the adiabatic efficiency. The injection of water into the compressor causes a variation of both parameters. As a result of cooling of the working medium, the process shifts toward isothermal compression, which is accompanied by a decrease in L_0 ($\bar{L}_0 = L_0/L_{0ad} < 1$). At the same time, the variations of the velocities of air, the surface friction (because of the presence of liquid films), and the emergence of additional loss of energy for providing the motion of liquid cause a variation of the internal efficiency, as a rule, toward decreasing ($\bar{\eta}_{0i} = \eta_0/\eta_{ad} \le 1$). The resultant effect is defined by the ratio $\bar{L}_0/\bar{\eta}_{0i}$. In experiments, it is difficult to determine the degree of variation of each one of these two parameters, and the resultant effect of injection is most often described by the unified effective efficiency η_{eff}. In so doing, this coefficient naturally loses its primary physical meaning as the measure of correlation between the ideal and real works of compression. Owing to the prevailing effect of $\bar{L}_0$, the effective efficiency usually exceeds the initial efficiency η_{ad}.

Figure 5 gives the effective efficiency as a function of relative flowrate of injected water for TV-3-117 and AL-21F3 compressors compared to the data of experi-

ments of [4, 18] and calculations for the compressor of a GT-009 gas-turbine unit in [16]. The calculated and experimentally obtained values of $\Delta\eta_{eff}$ for AL-21F3 (curves *1*, *2*, and *3*) agree well with one another up to $x_{water} \approx 1.5\%$. The difference may be due to the fact that, as is observed in [4], hydraulic loss in the inlet channel increased during the experiments because of the presence of a manifold of air-atomizing nozzles and because of extraction of compressed air for these nozzles. According to the estimates given in [4], these factors (which were not included in our calculations) may cause a reduction of effectiveness of compression by approximately 3.0–3.5% with $x_{water} = 2.0$–2.5%. With a further increase in the flowrate of injected water, the slope of the experimentally obtained dependence $d\Delta\eta_{eff}/dx_{water}$ is approximately halved, while the calculated curve retains the linear pattern; at $x_{water} = 2.5\%$, the experimentally obtained value of $\Delta\eta_{eff}$ is approximately 4.6%, while the calculated value is approximately 7.4%.

The discrepancies between the values of $\Delta\eta_{eff}$, especially for high values of x_{water} (2.0–2.5%), may be further caused by the fact that energy losses for changing the direction of the liquid phase motion are underestimated in calculations; in particular, the loss is ignored that is associated with the motion of films on the surfaces of the blades in the radial direction under the effect of centrifugal forces (which results in this loss being understated by a factor of approximately 1.5–2.0).

Comparison of calculated values of the increase in the effective efficiency of AL-21F3 and TV-3-117 compressors (see Fig. 5, curves *3* and *4*) reveals that the value of $\Delta\eta_{eff}$ of the latter compressor is approximately half that of the former. The main reasons for this difference include an order-of-magnitude-lower air flowrate and smaller standard size of TV-3-117 compressor compared to those of AL-21F3, as well as an almost 2.5 times higher speed of rotation in close-to-rated modes and an appreciably lower value of compression ratio.

Note that the increase in the effective efficiency of compressor by 3–5%, observed in the experiments of [18] and in our calculations, as well as a simultaneous certain increase in the actual flow rate of the working medium and in the compression ratio, must cause an overall increase in the GTU efficiency by 8–13% (relative) and an even more significant rise of this efficiency during GTU operation in combined cycles, in particular, in the case of "wet" compression and heat recovery, i.e., with simultaneous injection of water at the compressor inlet and into compressed air in the cycle with heat recovery [6].

The results of calculations for the GT-009 compressor given in [16] differ appreciably from the experimentally obtained values of $\Delta\eta_{eff}$ in the AL-21F3 compressor with parameters that are fairly close to those of the GT-009 [4]. According to [16], the calculated effective efficiency of the compressor increases by not more than 1 or 1.5%, with a maximum observed at low values of $x_{water} \approx 0.3$–0.6%. These results disagree with the results of other known experimental studies, for example, [3], where an increase in η_{GTU} by approximately 9 or 10% (relative) was obtained as a result of "wet" compression for the speed of rotation of the GT1500 engine of 8100 rpm/min and an airflow rate close to that in the AL-21F3; this corresponds to an increase in $\Delta\eta_{eff}$ of the compressor by approximately 2.5–3%.

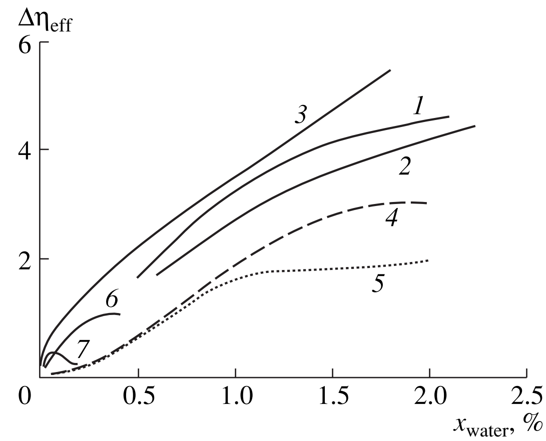

Fig. 5. The variation of $\Delta\eta_{eff}$ of a compressor as a function of the amount of injected water related to the airflow rate. AL-21F3 compressor: (*1*) delivery of water to the third guide-vane row, (*2*) delivery of water to the inlet header (experiments of [18]), and (*3*) delivery of water to the inlet header (our calculations). TV-3-117 compressor: injection upstream of the compressor (our calculations) (*4*) in view of the return of water from the casing, (*5*) the same, disregarding the return of water from the casing. GT-009 compressor (calculation of [16]): (*6*) injection to the third guide-vane row, (*7*) injection upstream of the compressor.

A possible reason for the small increment of η_{eff} in the GT-009 [16] is the relatively low value of π_{comp}, as a result of which the intensity of evaporation of moisture, especially in the high-pressure part of the compressor, may be appreciably lower. In addition, a significant reduction of $\Delta\eta_{eff}$ may be caused by the concentration of moisture in the vicinity of the compressor casing (see [16]) as a result of the shift of films on the rotor blades under the effect of centrifugal forces and separation of large (40–50 μm) droplets, as well as by the insufficiently high initial value of η_{ad} of the compressor.

The "wet" compression was calculated for the case of injection of water at the inlet to axial-flow compressors. As was revealed by the experiments of [4, 18], the injection into the third-stage guide-vane rows (at least for compressors with parameters close to those of the AL-21F3) is optimal from the standpoint of attaining the highest effectiveness of injection and the maximum of $\Delta\eta_{eff}$, although the injection of water at the compressor inlet lose very little in effectiveness as compared to the injection into the third-stage stator. The injection of water into the subsequent stages of compression (even

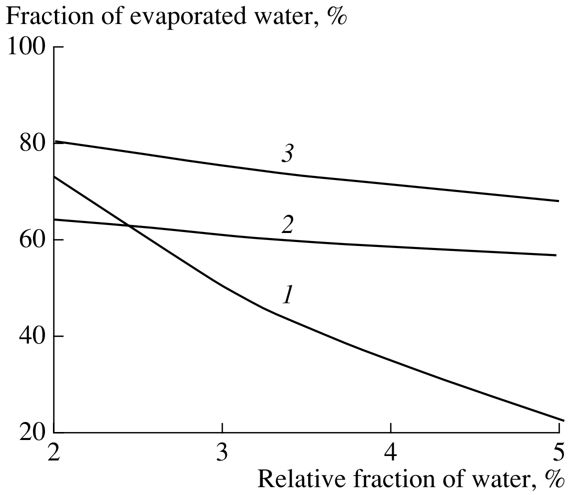

Fig. 6. The variation of the fraction of evaporated water as a function of the flow rate of delivered water for different droplet sizes d_{32} for TV-3-117 compressor; d_{32} in μm: (*1*) 25–20, (*2*) 3.0, and (*3*) 2.7.

the injection distributed over a number of stator stages, as well as via rotor blades) was less effective.

Analysis of the results of calculations of the behavior of dispersed droplets and films of water on the stator and rotor surfaces reveals that the deposition of droplets on the blades with subsequent shift of moisture under the effect of centrifugal forces toward the compressor casing is largely caused by the rather large size of the droplets, both of the droplets delivered from the nozzles (20–40 μm) and of those formed as a result of repeated fragmentation during the entrainment of films from the stator and rotor blades (8–15 μm). In so doing, the size reduction of droplets delivered from the nozzles even to 8–10 μm (for example, when using the Mee-fogging technology [20]) does not cause a significant change in their behavior in the second and subsequent stages of compression. The reason for this is the already mentioned separation of almost all initial droplets on the rotor blades even at the first (after the point of injection) stage of the compressor, while the formation of secondary droplets is defined by the thickness of water films entrained from the edges of the stator and rotor blades.

The processes of "wet" compression for the TV-3-117 with the droplet size reduced significantly was calculated in order to assess the impact made by the initial size of droplets. As a result, it was found that, if the fraction of evaporated water is 20–30% in the case of injection of water with droplets sized 15–20 μm and flow rate $x_{\text{water}} = 4$ or 5% (see Fig. 6), this fraction starts increasing abruptly (by a factor of 1.5–2) when the droplet size becomes 3–3.5 μm and smaller; at $d_{\text{drop}} \approx 2.5$ μm, more than 80% of injected water evaporates, with a corresponding increase in the effectiveness of "wet" compression. This occurs under the effect of two factors: first, a significant (four- or fivefold) increase in the overall surface area of droplets and of the intensity of their evaporation and, second, an almost complete absence of separation of droplets to the blade surface, with subsequent transition of water to the casing, although some wetting of the surfaces with the formation of very thin films is most likely retained. However, the reaction of these films to the impact of centrifugal forces and friction forces on the side of the airflow is much weaker. As a result, the losses of energy for acceleration and deceleration of liquid films on the blade surfaces decrease abruptly at the same time; this further promotes significantly the increase in the effective compression efficiency. Therefore, it appears very promising to finely atomize water during its injection to the compressor with the formation of droplets up to 2.0–2.5 μm in size.

It follows from the foregoing that the effectiveness of water injection in the case of "wet" compression may be increased not so much by way of optimization of the point of injection and of the amount of injected water as by employing injection techniques which would provide fine atomization of injected water in the form of a mist with droplets of prevailing (effective) size d_{32} of 2.0 μm or less. In view of the considered features of "wet" compression in aircraft engine-derivative GTUs characterized by a high speed of rotation and small size of axial-flow compressors, it appears most expedient to use the injection of microdroplets in application to such machines. The technology of injection of fine droplets (SwirlFlash technology) was developed by the Dutch company Kema [19]. This technology consists in using superheated water for injection via centrifugal nozzles. Water being atomized has a high temperature of 150–250 °C before the nozzles; in the case of an abrupt pressure drop from 10.0–25.0 to approximately 0.1 MPa during the passage through the nozzle channels, water in the volume of droplets formed at the outlet boils up intensely. As a result, according to [19], each droplet "explodes" under the impact of vapor being formed and is reduced in size to smaller droplets: with the initial diameter of droplets of about 20 μm, they are divided into thousands of droplets with an average size of approximately 2 μm.

Note that, although the SwirlFlash technology was both developed and tested in a 400-kW gas turbine, numerous questions associated with the uses of this technology remain open. In particular, it is necessary to know how the dispersion characteristics of the droplets being formed depend on the initial parameters of water, geometric shape of nozzles, and on other factors; whether the resultant two-phase flow in a real situation actually flows over the stator and rotor blades without the separation of moisture on their surface; how fast the variation of the droplet temperature is; and so on.

In order to study the "explosive" mechanism of division of droplets of superheated water injected into an air flow with parameters characteristic of the inlet sections of axial-flow compressors of a GTU, an experimental facility has been designed and manufactured at

the Joint Institute of High Temperatures, Russian Academy of Sciences (IVTAN), and is presently in the stage of adjustment. It is planned to use this facility for the investigation of the dispersion characteristics of droplets and of the distribution of their concentrations in an air flow, with the initial parameters of injected water varied in a wide range, namely, pressure up to 15 MPa and temperature up to 250 °C. A diagnostic system has been developed and adjusted that is based on determining the intensity of small-angle scattering of laser beam. A special CCD matrix is used for recording a large volume of information about the intensity of laser beam transmitted through the two-phase medium being investigated and deflected through different small angles. The control of the experiment and the processing of signals are fully computerized. Because droplets may be simultaneously observed in experiments in a wide range of variation of their sizes, namely, from initial 20–30 μm to 2 or 3 μm as a result of division of the droplets, it is planned to perform measurements at different wavelengths of diagnostic radiation.

It must be emphasized that the results of theoretical investigations of the behavior of the moisture phase under conditions of injection of water into high-speed and compact compressors of aircraft engine-derivative GTUs and the experimental data demonstrate the rationality of accomplishing the processes of "wet" compression in machines of electrical power of the order of 1 MW and higher and with a speed of rotation up to 15000 rpm. The effectiveness of "wet" compression increases when the flow rate of injected water increases to 2% of the airflow rate, and the increment of the effective efficiency of the compressor compared to that of its operation in a similar mode but without injection of water may amount to 2–4.5%. A reduction of the standard size of compressor (airflow rate and geometric dimensions) from the level of the AL-21F3 to that of the TV-3-117 with a simultaneous more than twofold increase in the speed of rotation causes a reduction of the fraction of evaporating water. It is interesting to note that, in spite of the foregoing differences in the speed of rotation and size of compressors under consideration, one can see in Fig. 1 that the moisture balances in these compressors differ insignificantly, although the evaporation of water in the AL-21F is somewhat higher. One of the main reasons for this phenomenon appears to be that, as is demonstrated by estimates, the average values of the integral of centrifugal radial accelerations of the rotor blades, equal to $0.5\,\omega^2(R_{cs}^2 - R_{sl}^2)$, where ω is the frequency of rotation and R_{cs} and R_{sl} denote the maximal (at the casing) and minimal (at the sleeve) radii of rotation of the blades, for both compressors differ relatively little. This parameter (which is selected in the structure proceeding from the permissible mechanical stresses in the blades) defines mostly the deformation of velocity fields and thickness of water films and, which is most important, the separation of moisture toward the casing; therefore, the approximate constancy of this parameter results in close correlations in the moisture balance under conditions of "wet" compression. The expediency of injecting water into machines of still higher speed of rotation and with still lower flow rates and smaller overall dimensions appears problematic. According to the estimates, one should expect a significant increase in the effectiveness of injection of water for small-size aircraft engine-derivative GTUs when employing fine atomization of injected water to droplets sized 2.5 μm or less. Data on the dispersion characteristics of injected water in the case of fine atomization are required for performing a reliable calculation of such "wet" compression.

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